CSA Engineering, Inc.

SIMULATION OF THE ZERO-GRAVITY ENVIRONMENT
FOR DYNAMIC TESTING OF STRUCTURES[1]

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ABSTRACT

Simulation of unconstrained (free-free) boundary conditions is a longstanding problem in ground vibration testing of spacecraft. The test article weight must be supported without introducing constraining forces due to stiffness, inertia, or friction from the suspension system. High-fidelity simulation of the space environment requires that such constraint forces be kept small compared to forces inherent in the experiment. A multipoint, six degree of freedom suspension system for dynamic testing is described. Intended primarily for highly flexible space structures, it uses a combination of passive pneumatic and active electromagnetic subsystems. The suspension offers a wide payload range, near-zero stiffness, zero static deflection, small added mass, and zero friction. The electromagnetic system can also provide active cancellation of added mass, accurate ride-height control, and integrated disturbance input. Several versions of the system are described, aimed at test articles ranging from very flexible solar arrays to a 7000-lb simulated optical truss. The concept and hardware are described, test results are given, and applications experience from several industry, government, and university installations is discussed.

INTRODUCTION

A primary characteristic of the space environment is the lack of gravity. This leads to a fundamental problem in space simulation for dynamic testing of structures. The weight of the test article must be supported in such a way that no significant constraining forces are imposed at the support points. Such constraints would change the normal modes of the test article and thereby produce unrealistic dynamics. Mathematically, the requirement is accurate simulation of so-called ``free-free'' boundary conditions.

Free-free conditions can be accurately simulated if the forces of constraint from the suspension system are small compared to internal forces due to stiffness and inertia. A common rule-of-thumb is that the vertical plunge frequency on the suspension should be at least an order of magnitude below the first flexural natural frequency of the test article. For natural frequencies above about 20 Hz, this requirement can often be met by hanging the test article from simple linear springs. For large or very flexible test articles with first frequencies below about 10 Hz, this simple approach becomes impractical due to static sag and surge modes of the springs themselves. As test article frequencies approach 1 Hz or less, the suspension problem becomes quite challenging. The most difficult cases occur when rigid-body dynamics (zero-frequency modes) are of interest.

Dynamic testing of low-frequency structures under simulated zero gravity requires a specialized suspension system. This paper describes such a system in several variants. Begun in 1988 as a research topic under the NASA/LaRC Dynamic Scale Model Technology Program [1] [2], the development has produced systems now in daily use on a number of aerospace programs as well as several specialized spin-off products. Called the pneu-mag system because of its combination of passive pneumatics and active electromagnetics, it provides a unique combination of wide payload range, very low stiffness, low added mass, and zero friction. These allow high-fidelity space simulation in a wide variety of applications.

The following section describes in general terms the requirements and figures of merit for a suspension system for zero-gravity simulation. The operating principle, analytical model, and current hardware implementation of the CSA Engineering pneu-mag system are described. Performance test results are given for comparison to the figures of merit. Finally, some recent developments for specialized applications are reviewed.

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SUSPENSION SYSTEM REQUIREMENTS

The essence of the suspension problem is to support the weight of the test article without imposing constraints or dynamic forces at the support points that would change its normal modes. Constraint forces can be produced by stiffness, mass, friction, or resonances of the suspension devices. The figures of merit described below are primarily measures of the degree to which these undesired forces are minimized.

Historically, efforts to develop hardware for zero-G simulation have often concentrated simply on achieving very low stiffness with adequate payload capacity. However, experience in the development reported here has shown that the low-stiffness requirement is only one of several and is not necessarily the most difficult or even the most important. A particular example of neglecting other effects in order to minimize stiffness has been the use of helium-filled balloons to suspend a test article. While vertical stiffness is small (zero were it not for density gradients in the surrounding air), it can easily be shown that the mass added by the balloons can never be less than about 16\% of payload mass. This added mass will corrupt even the lowest modes of the test article and will fundamentally change the higher modes due to inertia constraint forces. Further, the added mass cannot be accurately included in the analytical model because both it and its coupling to the article are anything but rigid. A systematic, balanced approach is necessary that considers all sources of undesired constraint forces.

Suspension systems for dynamic testing have usually taken the form of support-from-above cable arrangements with a soft spring in series with each cable. Pendulum action gave a simple, effective soft restraint in the horizontal direction, given adequate cable length. Two versions of the pneu-mag device described here use this overhead arrangement. Two others support their payloads from below, thus removing the headroom requirement and pendulum effects.

The remainder of this section describes specific requirements and how they have affected the development of the pneu-mag system.

Payload Applications to date have required from 15 to 600 lb per support point (i.e., per suspension device) for overhead suspension and as much as 3000 lb per device for support-from-below devices. While there is obviously no limit to what might be required, most applications seem to fall between 30 and 400 lb per support point. Since the suspension system is always soft relative to the payload, there is no limit to the number of devices that can be ganged to support a single payload: static indeterminacy is not a disadvantage. Of practical interest is the range of payloads that can be supported by a given system. The pneu-mag devices described here can typically accommodate a weight range of over 6/1 by immediate on-the-spot adjustments (no changing of parts) and about 20/1 with simple changes.

Static Deflection Linear springs, regardless of their implementation, are usually ruled out for low frequency applications by the contradictory requirements for low stiffness and small static deflection. A primary feature of pneumatic springs, shown later, is that they decouple the local tangent stiffness from static deflection. Pneu-mag devices use this feature to operate at zero static deflection.

Stiffness Allowable stiffness is usually specified in terms of the vertical rigid-body plunge frequency on the suspension. For a first flexural frequency of 1.0 Hz and plunge frequency of 0.1 Hz, this translates to approximately 0.1 lbf/inch of stiffness per 100 pounds of payload. Linear springs are obviously excluded; the static deflection would be 1000 inches. Figure 1 shows a more payload-specific
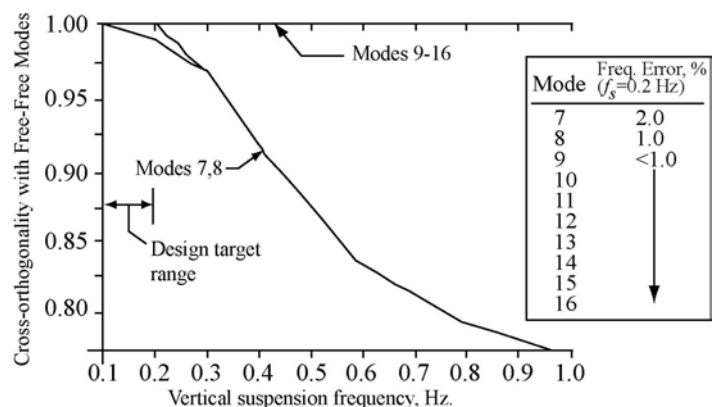
Figure 1   Effect of suspension stiffness on cross-orthogonality between unconstrained and suspended modes of a mass-loaded truss

method of determining suspension stiffness effects. It shows the gradual loss of cross-orthogonality between true free-free modes and modes including suspension stiffness as the suspension stiffness is increased. The example structure is a dynamic scale model of an early configuration of the space station. Physically, it is a number of rigid masses held together by light trusswork. Modes 7 and 8 are the first global flexing modes at slightly over 1 Hz. The example shows that the 10/1 frequency separation criterion is, in this case, probably conservative. It also shows that the effect of suspension stiffness varies greatly from mode to mode. The pneu-mag devices have been designed from the outset to have a variable suspension frequency with a range extending down to 0.1 Hz for most payloads.

Moving Mass Any suspension device adds some amount of mass to the payload, and can thus change its properties regardless of suspension stiffness. For very light, flexible payloads, it often occurs in low frequency testing that inertia constraint forces due to moving suspension mass exceed stiffness constraint forces for frequencies over a few Hz. Allowable added mass for the example structure described above was determined by finite element analysis to be about 5\%. The pneu-mag device easily meets this for payloads towards the upper end of its range. For very light payloads, an active mass canceling feature has been developed. Described in a later section, it has demonstrated the ability to cancel about 85\% of the 3.4-lbm moving element of a pneu-mag device with good stability margin. This allows testing a 20-lb payload with less than 3\% added mass.

Stroke Modal testing of even large spacecraft typically requires only a modest amount of vertical suspension stroke, on the order of one inch. The real requirement is set by measurement of local modes of very flexible appendages such as solar arrays, or by other dynamic tests involving rigid-body rotations of the payload. To meet these requirements, the current generation of pneu-mag devices has a vertical stroke of 6.0 inches. A large system concerned with rigid body modes has a stroke of over 10 inches.

Friction This is perhaps the most difficult requirement and has been an important driver in developing the pneu-mag concept. Unlike forces due to suspension stiffness or moving mass, friction is inherently nonlinear, often not repeatable, and always expensive to model in structural dynamics. A high priority was therefore given to reducing friction to the point where it had no measurable effect. This leads to a requirement that can be formulated in two ways. Either the friction force in the device must be small compared to its stiffness force, or friction force must be too small to produce a measurable change in payload acceleration based on typical accelerometer noise floor levels. Either leads to an allowable friction force on the order of 0.01\% of payload weight. The nominal specification for the pneu-mag devices was set at 0.005\%.

Controllability The projected large, flexible spacecraft applications that motivated the pneu-mag suspension system would have required 10-20 support points to avoid excessive weight-induced stresses. Even then, weight distribution between devices would have to be carefully controlled by tuning the ride height of each device. Testing efficiency required that this be done remotely, probably by a computer. While no such grand-scale application has actually materialized, the need for remote control was basic to the pneu-mag concept. The feature has proven to be valuable in day-to-day testing of smaller test articles. It has become a de facto requirement that a single operator be able to adjust from one location all devices of a multipoint suspension system in order to obtain a desired weight distribution.

Local Modes Accurate modal testing demands that the suspension devices contribute no local modes of their own within the test frequency band. Failing this, the suspension becomes part of the test article, a serious complication that the test engineer would much rather avoid. The pneu-mag devices are designed such that their only moving part is relatively light and stiff, with its first free-free mode well over 100 Hz.
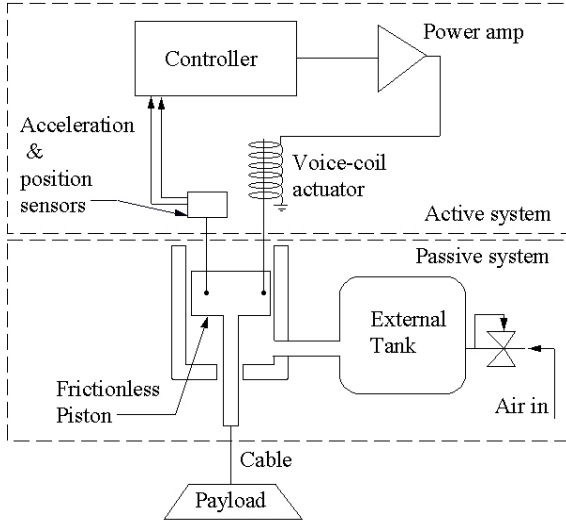
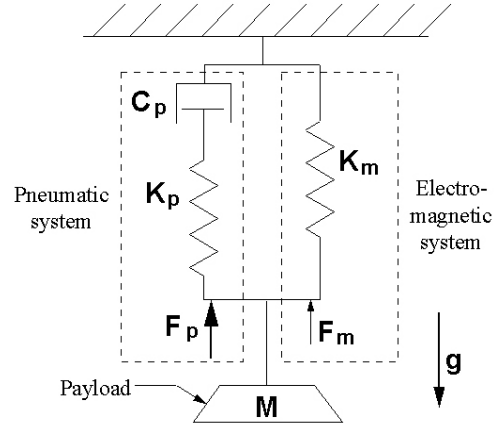Figure 2   Principle of pneumatic-magnetic suspension device

Figure 3   Simplified analytical model of pneu-mag suspension device.

PNEUMATIC-MAGNETIC PRINCIPLE

Figure 2 shows the basic operating principle of the pneu-mag device. Two parallel subsystems, one pneumatic and one electromagnetic, give the device its name. Figure 3 shows a somewhat simplified version of the analytical model for the device. In the figure, the constant forces F_p and F_m represent actions of the pneumatic air spring and magnetic fine-trim control respectively. The stiffness K_m is produced by the active system and K_p by the pneumatic system. The dashpot C_p is produced by the action of the pressure regulator.

The entire payload weight is borne by a special frictionless air piston operating vertically in a closely fitted cylinder. By porting the cylinder to an external volume through a large diameter line, the stiffness of the air spring thus formed can be made very small while retaining a large payload capacity. A precision pressure regulator maintains the mean cylinder pressure and supplies makeup air. For small volume changes, the regulated air spring behaves like a linear spring in series with a dashpot (Figure 3). The spring stiffness can be determined from elementary thermodynamics by assuming isentropic pressure changes. The result, expressed in terms of vertical suspension frequency, is as follows.

$$f_s = \frac{1}{2\pi} \sqrt{\frac{\gamma g A}{V}} \sqrt{1 + \frac{p_{amb}}{p_g}}$$

where f_s =suspension frequency, Hz

  γ =specific heat ratio for air (1.40)

  A =piston area, in^2

  V =tank volume, in^3

  p_{amb} =ambient pressure, psia

  p_g =gage pressure in air spring, psig

Eq. (1) and Figure 3 illustrate two well-known advantages of air springs. For p_g of several atmospheres or more, is f_s nearly independent of payload weight. That is, the suspension stiffness automatically changes in proportion to payload. Secondly, by adjusting the cylinder pressure to equilibrium with the payload weight, there is no static deflection of the spring.

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The series dashpot is the result of the action of the regulator. It vents air in or out of the external tank, attempting to keep the pressure constant in spite of piston movement. The spring-dashpot combination has a frequency-dependent stiffness with a first-order, high-pass break frequency determined by the size of the external tank and the characteristics of the regulator. The regulated pneumatic spring has no static stiffness at all: it can be in equilibrium with the payload weight at any vertical position of the piston within the cylinder. This property can be a disadvantage since the payload can float to the end of the stroke and strike the limit bumper. Thus arises the need for the parallel magnetic system.

The active part of the system includes a custom long-stroke, voice-coil magnetic actuator, a displacement sensor, controller, power amplifier, and (optionally) an acceleration sensor. A negative-feedback, low-gain displacement loop is implemented to serve as a noncontact "spring" with a very small, variable stiffness. It keeps the moving element of the device near the center of the working stroke, thus preventing it from striking the limit bumpers when excitation is applied to the payload. A secondary use of the displacement loop is to provide fine-trim control of ride height. This is required if the payload contains optics that must be kept in registration with some off-payload datum. A small, variable DC offset voltage can be injected into the displacement loop. It serves the same purpose as adjusting the pressure regulator set point but does so with much faster response and much greater resolution.

The active system also provides other useful features. A positive-feedback acceleration loop can be added to cancel most of the moving mass of the suspension device. This can drastically improve the fidelity of the zero-G simulation for small payloads, on the order of 20 pounds, where the added mass can have a significant effect [3]. An external dynamic signal can be summed into the loop to provide a controlled vertical excitation to the payload. This is useful in situations where the payload is very flexible and would be affected by the rotational constraint of a shaker pushrod. A modest multipoint-excitation modal test can be performed using only the actuators and sensors built into the pneu-mag devices. The DOF's are, of course, limited to the vertical direction at the hang points.

Special versions have been built with a lowpass filter in the displacement loop such that DC stiffness could be increased for improved centering without imposing constraining stiffness at higher frequencies. Negative velocity feedback was necessary to stabilize the loop in the presence of the phase lag of the filter. The arrangement was not entirely satisfactory because the velocity feedback slightly increased the apparent damping of the lowest modes of the test article [4]. The eventual solution was a basic redesign with much longer stroke (described below) which eliminated the need for enhanced DC stiffness. Frequency-dependent stiffness is one of many features that have been investigated in a research context during the pneu-mag systems development.

HARDWARE IMPLEMENTATION
Passive / Active System
Figure 4 shows two views of a current pneu-mag version along with nomenclature. Figure 5 shows the pneumatic and electronic control panels for a system at NASA Langley
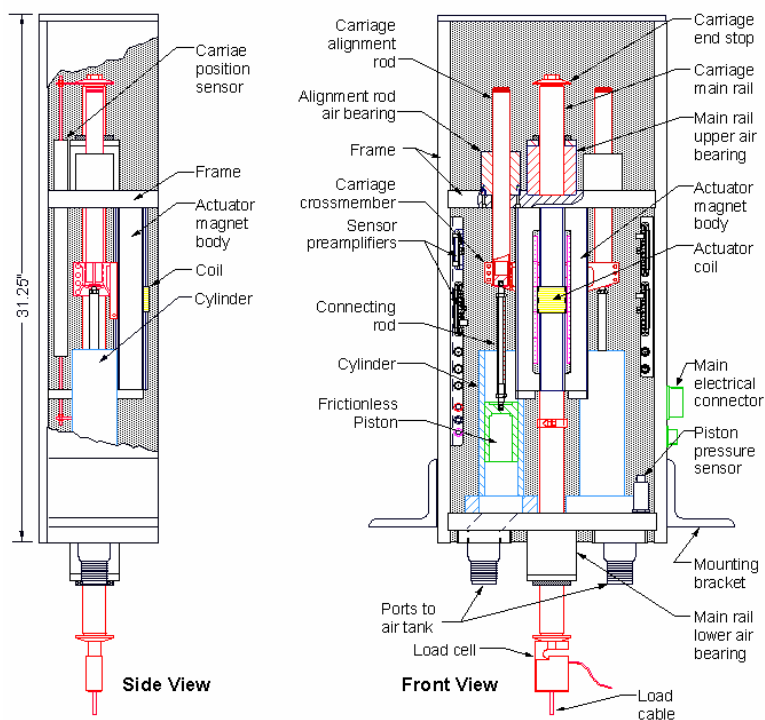

Figure 4   Passive-active suspension device currently in use at NASA/Jet Propulsion Laboratory (Microprecision Interferometer test bed) and NASA Langley Research Center (Large Structures Lab).

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Figure 5 Pneumatic and electronic control rack for 6-device suspension system

Research Center. Figure 6 shows several views of a system in use at NASA/Jet Propulsion Laboratory with the Microprecision Interferometer test bed. The active capabilities are particularly useful in this latter application because of the need to maintain accurate ride height. Optics on the suspended payload must receive the image of a synthetic star from an optical bench supported off the laboratory floor.

Referring to Figure 4, the pneu-mag device is composed of a carriage that moves vertically in a rectangular box frame on four air journal bearings. The moving carriage is composed of a single main rail down the center with a horizontal crossmember fixed to it about one-third of the way from its upper end. Two identical frictionless piston/cylinder sets are used, one on either side of the main rail, with each piston lifting against one end of the crossmember via a connecting rod. Both cylinders are ported through the bottom of the box frame to hoses connected to a common accumulator tank. The single-main-rail, twin-piston geometry is used to avoid bending stress in the rail and allow a high payload capacity (large piston area) with minimum carriage mass. The design replaced an earlier single-piston configuration with shorter stroke [4][5].

The payload is connected to the carriage by a cable that attaches to the lower end of the main rail. An optional load cell may be interposed between the cable and carriage to sense cable tension.

The magnetic voice-coil actuator is a custom, long-stroke design that uses a coil connected to the carriage crossmember and a magnet body mounted to the frame. High-energy NdFeB magnets are used for maximum force capacity. The magnet structure is designed to produce a high flux density in the air gap with minimal variation in the stroke direction.

The carriage displacement sensor is a non-contacting LVDT mounted behind the carriage main rail. A piezoresistive (DC-coupled) accelerometer is mounted to the carriage for the mass-canceling loop. A pressure sensor transduces the cylinder pressure to obtain a continuous analog reading of piston force. Each suspension device connects to the control panel by a single electrical cable and two compressed air lines.

The key to the performance of the device is the fact that the load is supported completely on air. The combination of air bearings, noncontacting actuator and displacement sensor, and floating pistons eliminates any source of friction. The only connections between the moving and fixed parts of the device are the small wires carrying current to the actuator coil and (optionally) the small signal cable from the accelerometer.

Nominal specifications for the device as shown are: maximum payload (at 80 psig): 500 lb; active stiffness: variable over 0.05-2.0 lbf/inch; vertical suspension
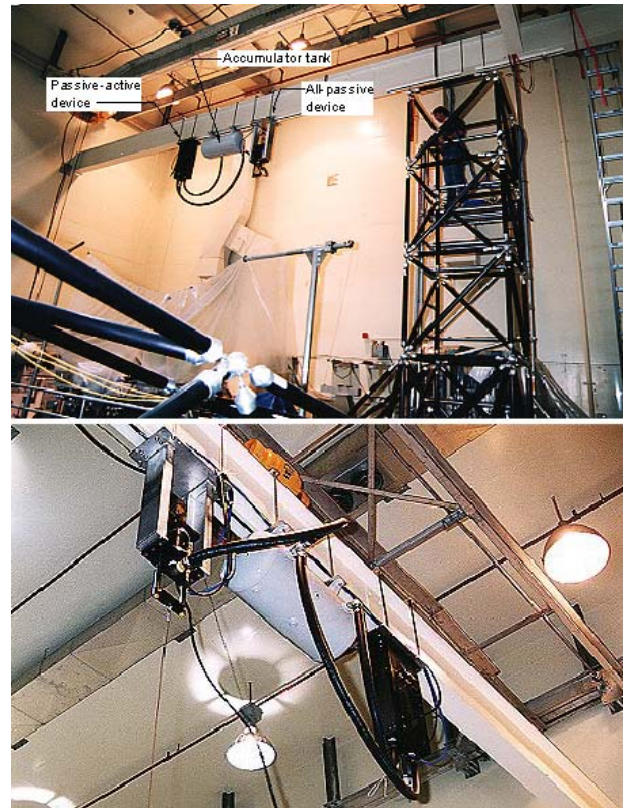

Figure 6 Zero-G suspension system being installed at NASA/Jet Propulsion Laboratory for the Microprecision Interferometer test bed. Top photo shows one passive-active and one all-passive device ganged to support the MPI optics boom. The two devices are connected to a common accumulator tank. Bottom photo is a close-up of the devices. The full system uses three passive-active devices and one all-passive.
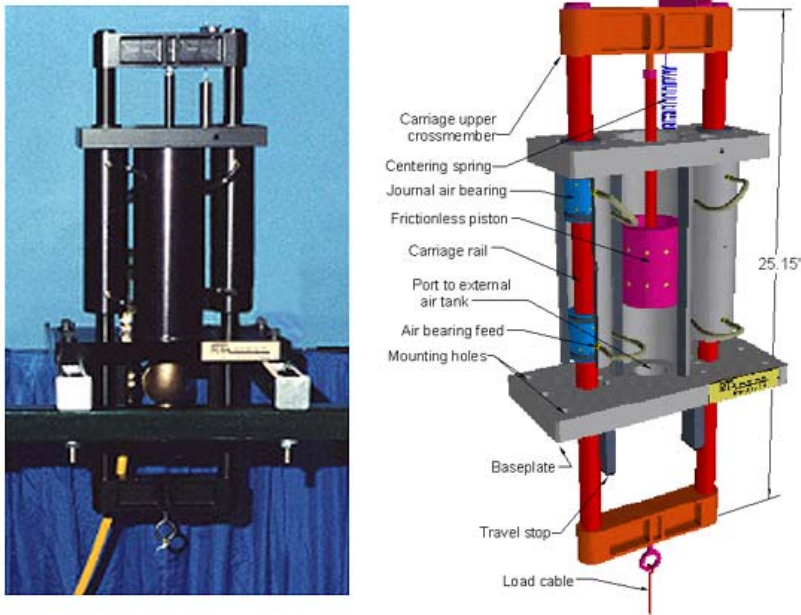
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Figure 7 All-passive version of suspension device

frequency: 0.1-0.2 Hz; breakaway friction: under 1 gram; stroke: 6.0 inches; carriage mass: 6.0 lbm.

All-Passive Device

Figure 7 shows a version of the device which substitutes a simple mechanical extension spring for the electromagnetic active system. This all-passive device came about in response to applications which would use the active system only for centering control and which did not require precision ride-height maintenance or other active features. The passive device has the advantages of being substantially simpler, cheaper, and lighter. It retains the high payload, low stiffness, long stroke, and zero friction attributes of its predecessor. Devices of both types can be mixed in a single system to utilize the advantages of both.

PERFORMANCE TEST RESULTS

This section presents some typical test results for pneu-mag suspension devices. A rigid test payload was used in all cases.

Figure 8 shows the measured vertical displacement/force FRF with two different payloads. The displacement loop was closed with a loop gain of about 0.2 lbf/inch. The internal actuator and displacement sensor were used to allow FRF measurements at the very low frequencies involved. The series-dashpot effect of the pressure regulator becomes progressively more evident as the payload is increased. At 350 lb, the suspension resonance has disappeared due to regulator action. This is exactly what is needed in a suspension device; the load is virtually floating with negligible vertical constraint.
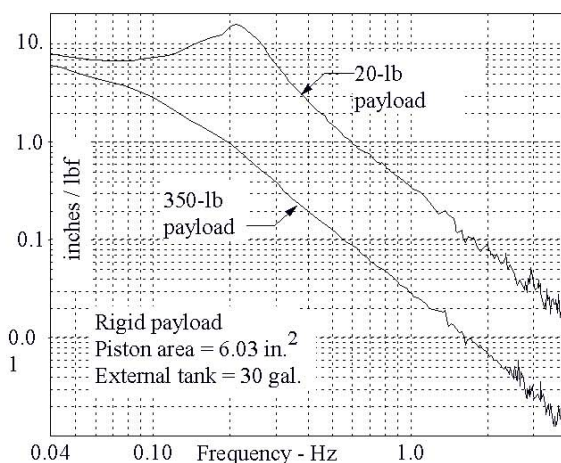

Figure 8 Vertical displacement/force frequency response function measured for passive-active device with rigid payload.
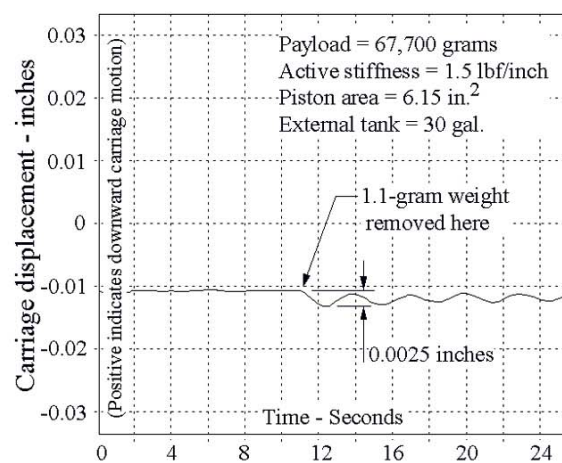

Figure 9 Typical breakaway friction test result for passive-active suspension device.

Figure 9 shows a typical breakaway friction test result. The pneu-mag device was loaded to 150 lb (68 kg), active loop stiffness was set to 1.5 lbf/inch, and a 1.1 gram weight was carefully lifted from the carriage while monitoring the position sensor output. Transient motion of the carriage, small but clearly visible against the quiet background, shows that even this tiny load step was more than sufficient to cause motion. Friction is too small to measure.

SPECIAL VERSIONS

The pneu-mag devices shown above are third and fourth generation designs. They were designed primarily around the requirements of modal (i.e., small displacement) testing of low frequency structures in a typical laboratory setting. Several more specialized variants have also been developed [6].

Large-Scale Pointing and Tracking Experiments

A variant of the pneu-mag suspension is in use at the Air Force Phillips Laboratory, Kirtland AFB, NM. Designed and built by CSA Engineering, it supports a large precision truss structure (Figure 10). The facility, simulating a large, orbiting optical system, has been developed for ground-based pointing and tracking experiments.

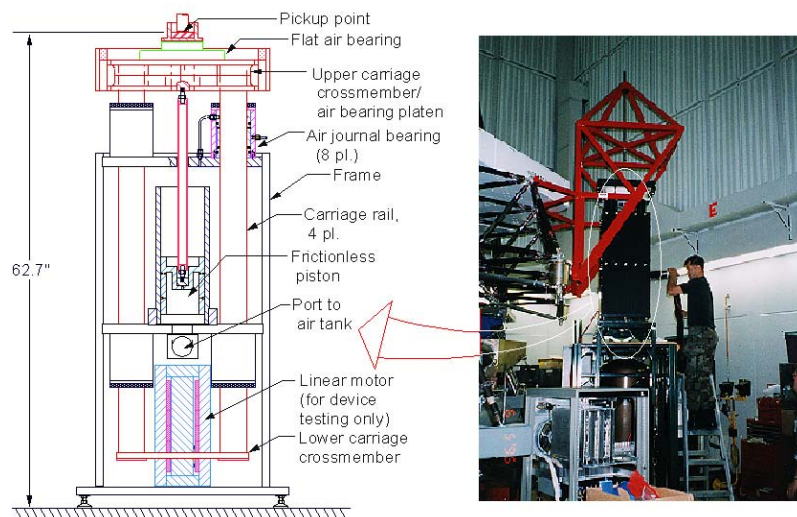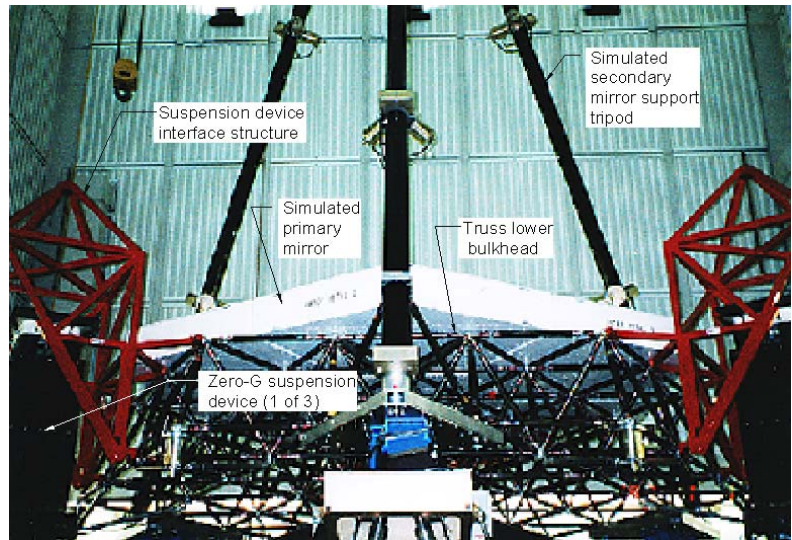The zero-G suspension consists of three identical devices, each supporting a hard

Figure 10   7000-lb truss testbed structure floated on three floor-standing zero-G suspension devices. Horizontal motion at each support point is allowed by a flat air bearing running on the ceramic upper crossmember platen of the device.

point of the truss. Each device uses a frictionless air spring and air bearing carriage similar to those described earlier. However the devices are much larger; each is capable of floating about 3000 lbs. Also, the system supports the payload from below. Horizontal motion at each support is accommodated by a flat air bearing sliding on a ceramic platen that forms the upper crossmember of the vertically moving carriage. This eliminates the horizontal stiffness and other limitations of support-from-above pendulum systems. Each support point can accommodate a vertical stroke of 10.5 inches and horizontal motion of 8.7 by 10.7 inches. Vertical plunge frequency is nominally 0.1 Hz and is heavily damped by the action of the air pressure regulator. As in the smaller versions, the system is completely frictionless over its full range of motion.

High-Performance Airborne Vibration Isolators

The very soft, frictionless character of the pneu-mag suspension suggests its application to vibration isolation. It is potentially capable of 1-2 orders of magnitude improvement in isolation relative to conventional passive systems. Also, it can provide a long stroke, easily 3 orders of magnitude longer than active piezoelectric-based systems. These properties make it well suited for airborne optics where precision beam pointing must be performed in the presence of significant broadband base vibration.
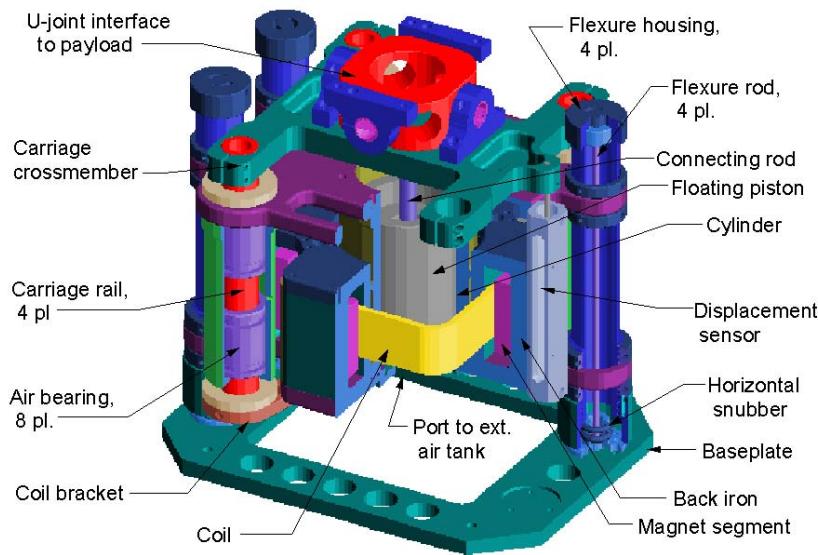
Figure 11   6-degree of freedom pneumatic mount with integral voice-coil actuator. The device is derived from zero-G suspension technology and is sized for payloads up to 1200 lb. per device.

A variant of the support-from-below pneu-mag system has been developed for such applications. A single suspension device is shown in Figure 11 and a full-scale demonstration testbed system is shown in Figure 12 with a simulated 1500-lb optical bench payload. Like the system described in the last section, this testbed has been designed and built by CSA Engineering for the Air Force Phillips Laboratory.

A unique feature of the airborne problem is low-frequency inertial forces on the isolated payload due to aircraft maneuvering. As the aircraft turns and banks, it will impose significant g loading on the payload. This would produce unacceptable static sag on a suspension designed for a very low break frequency. This situation is countered by the active system. The isolation mounts incorporate voice coil actuators and DC-coupled acceleration sensors. The actuators use rare earth magnets and air-jet coil cooling to produce a force rating of about 230 lbf from a 96-lb actuator with a 1.25-inch stroke. Similar actuators also act on the payload in the horizontal direction to provide inertia force compensation and damping of suspension modes. Various software-based control schemes are being investigated to optimize the performance of the active system for airborne isolation.
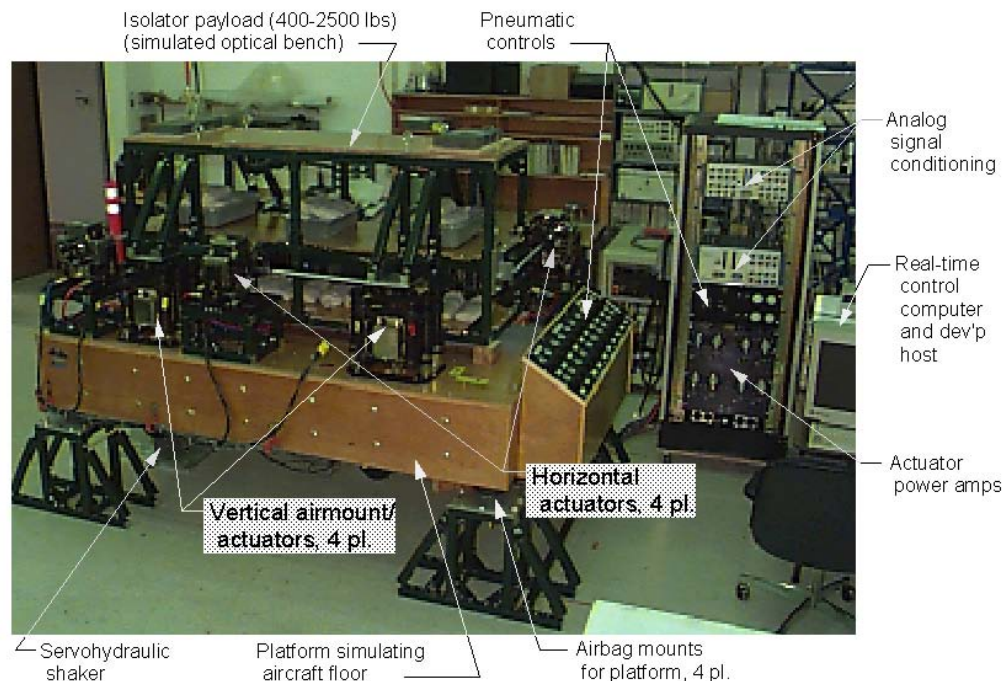

Figure 12   Demonstration testbed for passive-active isolators. The payload is a 1500-lb simulated optical bench. Enhanced active features are provided by a VMEbus real-time control computer.

CONCLUSIONS

A suspension system has been described for producing near-unconstrained boundary conditions in dynamic testing of flexible structures. Called the pneu-mag system, it provides a unique combination of high payload, low stiffness, low added mass, and zero friction. It has reached the status of a commercial product and is in daily use on several current aerospace programs. Work is continuing to develop new versions for specialized applications.

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